

# Numerical Analysis of Convection/ Transpiration Cooling

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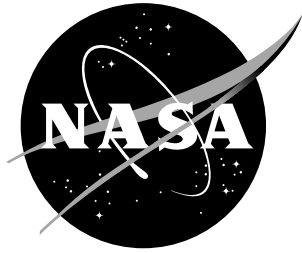
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## Abstract

An innovative concept utilizing the natural porosity of refractory-composite materials and hydrogen coolant to provide CONvective and TRANspiration (CONTRAN) cooling and oxidation protection has been numerically studied for surfaces exposed to a high heat flux, high temperature environment such as hypersonic vehicle engine combustor walls. A boundary layer code and a porous media finite difference code were utilized to analyze the effect of convection and transpiration cooling on surface heat flux and temperature. The boundary layer code determined that transpiration flow is able to provide blocking of the surface heat flux only if it is above a minimum level due to heat addition from combustion of the hydrogen transpirant. The porous media analysis indicated that cooling of the surface is attained with coolant flow rates that are in the same range as those required for blocking, indicating that a coupled analysis would be beneficial.

## Nomenclature

a	= coefficient in Nu
b	= coefficient in Nu
B <sub>o</sub>	= coefficient for first order effect in permeability
Cp <sub>c</sub>	= coolant specific heat
d	= coefficient in finite difference equation
D <sub>h</sub>	= hydraulic diameter of the coolant channel
G	= mass flow rate per unit area
h	= heat transfer coefficient
i	= spacial index in finite difference equation
k	= thermal conductivity
K	= permeability
K <sub>o</sub>	= coefficient for second order effect in permeability
L	= thickness of porous media
m	= molecular mass

M	= molecular weight
Nu	= Nusselt number
$\Delta P$	= pressure difference ( $P_{ch} - P_{comb}$ )
P	= pressure
Pr	= Prandtl number
q <sub>appl</sub>	= applied heat flux at combustor wall
Re	= Reynolds number
R <sub>u</sub>	= universal gas constant
R <sub>H2</sub>	= hydrogen gas constant
T	= temperature
V	= velocity of coolant
x	= spacial coordinate in porous media

## Greek

$\epsilon$	= porosity
$\kappa$	= Boltzmann constant
$\mu$	= viscosity
$\rho$	= density

## Subscripts and Superscripts

avg	= average
c	= coolant in porous media
ch	= coolant in coolant channel
comb	= combustion chamber
eff	= effective
m	= porous media
s	= surface
v	= volumetric, in porous media
wall	= conditions at surface of combustor wall

## Introduction

Design and trade studies of hypersonic vehicles have shown significant benefits for operation at high engine wall and coolant/fuel temperatures. One approach to achieving these operational conditions in engine combustors has led to the development of a platinum clad

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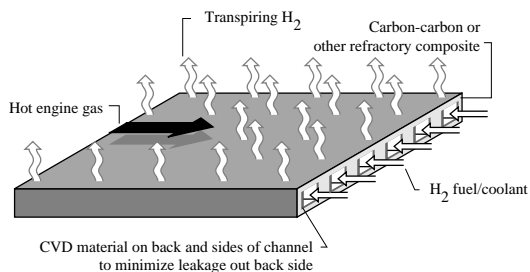
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Mo-50Re alloy for the convectively cooled structure. This approach has been taken because of the requirements for a material with both high-temperature stability and high thermal conductivity to accommodate the anticipated high heat fluxes. However, these materials are expensive and the cladding has not been reliable. Because Mo-Re has poor oxidation resistance, serious problems can result from exposure of the Mo-Re to oxygen should a breach occur in the cladding.

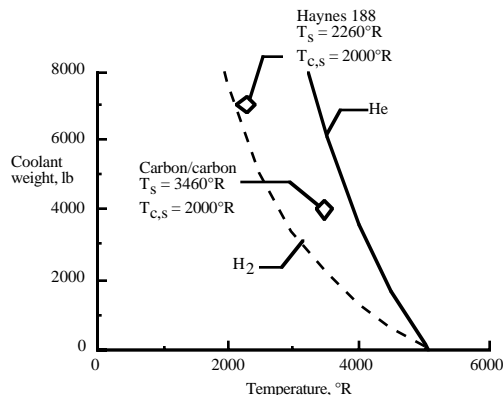
Cooled refractory composites have been considered for nozzles but have not been proposed for combustors because of their relatively low thermal conductivity. Furthermore, the use of convectively cooled refractory composites using leak-tight metallic liners in the coolant passage and oxidation protection surface coatings cause additional design constraints due to material incompatibility and thermal expansion differences between the tubes and the liners as well as the external coating compatibility with the combustor environment, respectively.



**Fig. 1. CONTRAN cooling concept.**

An innovative concept utilizing the natural porosity of refractory-composite materials and hydrogen coolant to provide CONvective and TRANspiration (CONTRAN) cooling and oxidation protection has been proposed for surfaces exposed to a high heat flux, high-temperature environment such as hypersonic vehicle engine combustor walls. Specifically, the concept relies on the hydrogen coolant to permeate the selectively densified refractory-composite material such as carbon/carbon (C/C) to provide a reliable cooling and oxidation protection system. The basic CONTRAN concept, which is shown in Fig. 1, is based on the fundamental premise that the natural porosity of refractory-composite materials leads to leaks. Though little can be done to completely eliminate the leaks, the leak rate can be controlled through selective densification of the refractory-composite materials. The hydrogen coolant which permeates the C/C prevents oxidation, and the transpiring coolant also significantly reduces the heat flux. Since the transpiration cooling is very effective in cooling the structure and it blocks some heat input exterior to the surface, the very high thermal conductivity requirements imposed by a pure convective cooling system no longer exist. Similarly, the needs for

impermeable oxidation protection coatings and leak free coolant liners are eliminated. An important aspect of the CONTRAN concept is control of the spatial variation of the hydrogen permeability through the composite, both to minimize the coolant leakage out of the back side and to accommodate the variations in the heating conditions over the hot surface.



**Fig. 2. Coolant weight as a function of temperature for hydrogen and helium coolant. Also shown are design conditions for Haynes 188 and C/C closure door.**

Potential benefits of the application of the CONTRAN cooling concept are illustrated in Fig. 2. The particular example shown is for an inlet closure door, not a combustor wall panel, and the cooling mode is pure transpiration (with an energy balance for which wall surface temperature equals outlet coolant temperature), which is a limiting case of the CONTRAN concept. The heat blocking effects of transpiration cooling are not considered. The closure door is used during re-entry to prevent hot gases from entering the engine, and the expendable coolant, which is dumped overboard, represents a direct weight penalty. The dashed line in Fig. 2 illustrates the reduction in coolant weight penalty as a function of hot surface temperature where the coolant temperature is permitted to rise to the surface temperature, as it will with transpiration cooling. If the surface is allowed to rise to  $\sim 5000^{\circ}\text{R}$ , no weight penalty is incurred since no coolant is required. The figure also presents coolant weight requirements using helium as the coolant medium to alleviate concerns about external burning of the hydrogen coolant. Previous engine contractors' design studies, as shown by the two design conditions (symbols), indicate that the coolant weight penalty can be reduced by  $\sim 3000$  lb by switching from the baseline Haynes 188 alloy door to a C/C door. However, the C/C design is constrained by the  $3460^{\circ}\text{R}$  limit imposed by the oxidation protection coatings and the  $2000^{\circ}\text{R}$  limit imposed by metallic tube liners. For the C/C design, an additional 2000 lb savings can be realized (as shown on the dashed line) with transpiration

cooling by the elimination of the 2000°R limit for the coolant tubes.

The lack of serious consideration of transpiration cooling for engine combustors may be ascribed to preconceived concerns about and lack of knowledge of the concept and the specific application for which it has been proposed. Transpiration cooling has been proposed for the engine cowl lip and other stagnation regions. However, there has been serious concern about the transpiring coolant altering the effective aerodynamic shape of the components, the potential detrimental effects of shock impingement on transpiration cooling effectiveness, and the unknown effects of combustion of the transpirant on the cooling process, as well as the potential impact of that combustion on the inlet and overall engine performance. Several of these concerns are addressed in paper.

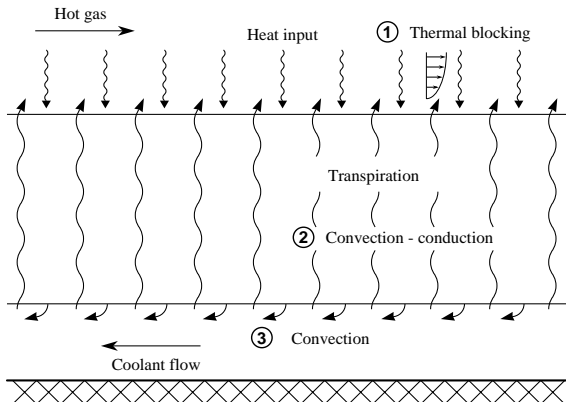


Fig. 3. Schematic diagram showing aspects of CONTRAN cooling to be modeled in computer analysis.

The numerical analysis results presented here utilized two codes to model the CONTRAN cooling of C/C. A boundary layer code was used to calculate the effect of the transpiration cooling on reducing the heat flux at the surface (effect 1 in Fig. 3). The porous media code modeled both the convective cooling in the coolant channel and the thermal-fluid mechanics of transpiration in the porous media, as illustrated by effects 2 and 3 in Fig. 3. The two codes, though not coupled, were used together to demonstrate the feasibility of using CONTRAN cooling for scramjet engine combustors.

## Boundary Layer Analysis

The viscous flow along the C/C surface was computed using an implicit finite-difference boundary layer code.<sup>1</sup> A binary diffusion model was assumed with injection of the hydrogen coolant into the freestream gas. The species composition of the gas mixture was determined using a Gibbs free-energy minimization

procedure. The external flowfield was modeled as a flat plate with either a constant or varying pressure at the boundary layer edge. Injection of coolant was initiated downstream of the leading edge of the flat plate to permit injection into a developed turbulent boundary layer. The heat transfer predicted by the code included both combustion and the blocking effect of the hydrogen coolant. In all cases, the boundary layer was assumed attached. Elimination or removal of oxygen at the surface resulted from burning of the oxygen with the transpiring hydrogen, not removal of the boundary layer.

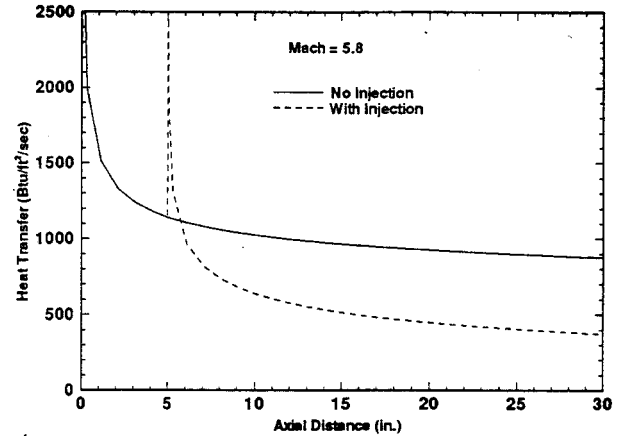


Fig. 4. Heat transfer rates on a flat plate with and without transpiration cooling.

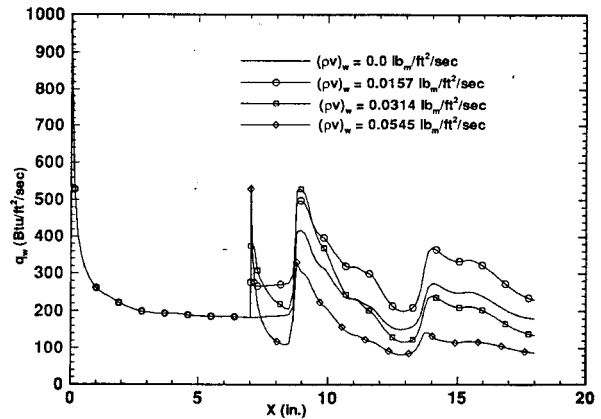


Fig. 5. Effect of hydrogen mass flow rate on the surface heat flux with a pressure gradient.

An analysis of a shock-free flowfield using the boundary layer code is shown in Fig. 4. The figure shows the heat transfer rates with and without hydrogen transpiration. The inflow conditions for this case (Mach 5.8) are based upon flight-scale scramjet combustor conditions. The geometry was a flat plate, and a turbulent boundary layer was developed before injection begins. The spike in the heat transfer with injection was caused by the immediate burning of the hydrogen coolant and surface oxygen when injection starts. After the

oxygen near the surface was eliminated by burning with hydrogen, the heating rate drops below the no injection case. This result demonstrates the blockage effect of the hydrogen transpiration.

The boundary layer code was also used to simulate hydrogen transpiration into a scramjet combustor flowfield with pressure gradients to simulate shockwaves. The inflow condition (Mach 4.22) was from a pulse facility which is used to test subscale scramjet designs. The shockwave was modeled by varying the pressure at the boundary layer edge. Heat transfer results from a parametric study of the injection rates are shown in Fig. 5. A sharp increase in pressure occurs at approximately 9 in., with a smaller increase at approximately 13.5 in. Injection of hydrogen starts slightly ahead of the first shock impingement. At the lowest injection level, the heat transfer to the surface had actually increased due to the hydrogen injection. As the injection rate increased, the heat transfer was reduced below the no injection case.

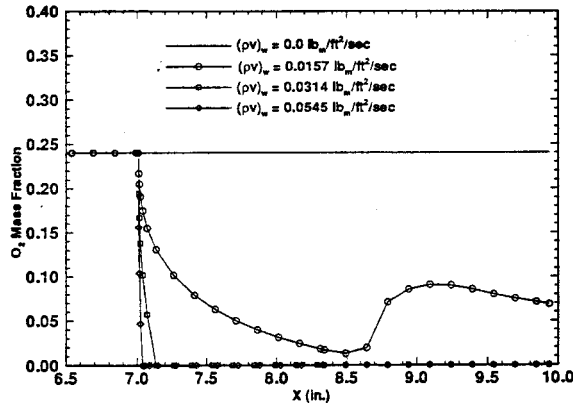


Fig. 6. Effect of hydrogen mass flow rate on the oxygen concentration at the surface.

The reason for the increase in heat transfer at low injection levels is shown in Fig. 6. This figure shows the axial variation of the oxygen concentration at the surface. At the lowest injection level, the oxygen was not eliminated from the surface and injected hydrogen continued to burn near the surface. This burning resulted in increased heating rates. At the highest injection level, the oxygen was rapidly removed from the surface and the heating rates decreased. In this case, the burning of the hydrogen occurred away from the surface and the expected reduction in heat transfer from transpiration cooling was realized.

The effect of combustion of transpirant on heat transfer has been validated by changing the composition of the inflow gas. The heating rates for air and pure nitrogen are shown in Fig. 7. The nitrogen inflow case does not have a spike in the heat transfer immediately

following the start of injection because of the absence of oxygen. In this case, the heat transfer drops immediately after the injection begins. After the first shock impingement, the results with the nitrogen inflow are slightly below the air inflow case.

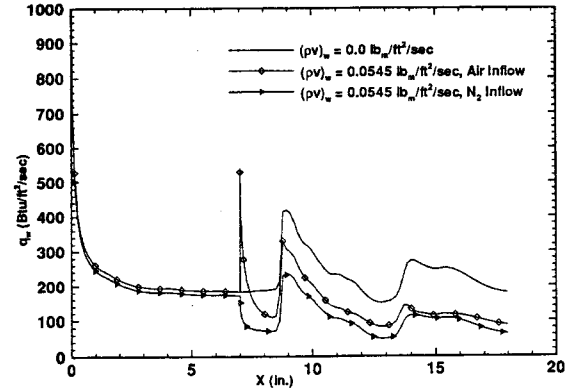


Fig. 7. Effect of inflow gas on the surface heat flux.

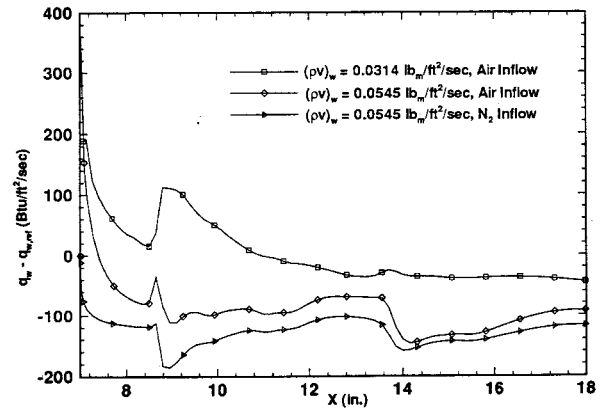


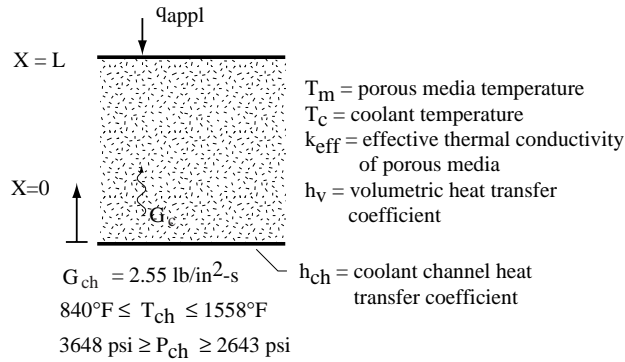
Fig. 8. Blockage effect of the gas injection on the surface heat flux.

The blockage effect of the hydrogen injection is shown in Fig. 8. In this figure, the difference in heating rate due to injection is plotted for several cases considered in this study. The effects of nitrogen inflow and increased injection rates are clearly shown. The boundary layer code indicates that the pressure gradients due to the shocks do not remove the transpiration flow from the surface and reduce the cooling. Instead, the region where the shocks impinge experience a significant cooling effect of the transpiration flow. These results confirm the ability of the CONTRAN concept to provide thermal and oxidation protection using hydrogen transpiration through a porous medium.



## Porous Media Analysis

The approach taken in the porous media analysis was to solve three coupled differential equations for the C/C temperature, the hydrogen temperature, and the hydrogen pressure in the porous media. A general convection boundary condition was assumed at both boundaries of the porous media, i.e., inside the coolant channel and at the combustor surface. At the combustor wall surface, blocking (a reduction) of the applied heat flux occurred due to the injection of the hydrogen into the boundary layer. The permeability and mass flow rate were calculated in the code. A discussion of the numerical procedure used for solving the three equations for the three unknown ( $T_m$ ,  $T_c$ ,  $P$ ) as well as the porous media (C/C) permeability and the coolant (hydrogen) mass flow rate follows.



**Fig. 9. Schematic diagram of the porous media and the nomenclature and channel coolant values used in the thermal analysis.**

### Porous Media Temperatures and Coolant Temperatures and Pressures

The first step was to calculate the porous media temperature and coolant temperature and pressure. This required the solution of three differential equations for three variables,  $T_m$ ,  $T_c$ ,  $P$ . A schematic diagram of the porous media with the channel coolant flow rate, temperature, and pressure is shown in Fig. 9.

A 1-D energy balance on the porous media results in the following equation:

$$\frac{d^2 T_m}{dx^2} - \frac{h_v}{k_{eff}} (T_m - T_c) = 0$$

where  $T_m$  is the porous media temperature,  $T_c$  is the coolant temperature,  $h_v$  is the volumetric heat transfer coefficient in the porous media, and  $k_{eff}$  is the effective thermal conductivity of the porous media as measured by treating the porous media as a porous solid with no coolant.

The volumetric heat transfer coefficient,  $h_v$ , is

calculated according to an empirical formula for flow in a porous media and has units of  $\text{Btu/in}^3\text{-s-}^\circ\text{R}$ . The porous media flow Nusselt number is defined by Florio<sup>2</sup> as

$$Nu_v = a Re^b$$

where  $a = 2.22 \times 10^{-6}$ ,  $b = 0.703$ , and the Reynolds number,  $Re$ , is defined as

$$Re = - \frac{\rho_c B_o^{1.5} dP}{\mu_c^2 \epsilon dx}$$

The coolant density is defined as  $\rho_c$  and the coolant viscosity is defined as  $\mu_c$ . The porosity is given by  $\epsilon$ . The coefficient  $B_o$  is the coefficient of the first order effect term in the permeability  $K$  (described subsequently).

The pressure of the coolant in the porous media is the second variable that must be obtained, and it is obtained from the solution of a first order differential equation given by

$$- \frac{dP}{dx} = \frac{\mu_c G_c}{\rho_c B_o}$$

where  $G_c$  is the mass flow rate per unit area and the viscosity and density are functions of temperature and pressure.

Once  $Nu_v$  is calculated,  $h_v$  is obtained from

$$h_v = \frac{Nu_v k_c}{B_o}$$

where  $k_c$  is the thermal conductivity of the coolant. The volumetric heat transfer coefficient,  $h_v$ , was calculated at each node as a function of temperature.

The boundary conditions are given as

$$h_{ch} (T_{ch} - T_m) = - k_{eff} \frac{dT_m}{dx} \quad \text{at } x = 0$$

$$q_{appl} = - k_{eff} \frac{dT_m}{dx} \quad \text{at } x = L$$

where  $q_{appl}$  is the applied heat flux at the boundary  $x = L$ .

The coolant heat transfer coefficient,  $h_{ch}$ , was calculated from the Nusselt number for turbulent flow. The coolant channel Nusselt number is defined as<sup>3</sup>

$$Nu_{ch} = 0.027 Re^{4/5} Pr^{1/3} \left[ \frac{\mu_{ch}}{\mu_s} \right]^{1/4}$$

where  $\mu_{ch}$  and  $\mu_s$  are the coolant viscosity at the temperature of the coolant in the coolant channel and at

the cold surface temperature of the porous media, respectively.

Once the Nusselt number was calculated, the coolant heat transfer coefficient was obtained from

$$h_{ch} = \frac{Nu_{ch} k_c}{D_h}$$

where  $D_h$  is the hydraulic diameter of the coolant channel.

The third equation to be solved gives the coolant temperature distribution. An energy balance on the coolant yields

$$\frac{dT_c}{dx} - \frac{h_v}{G_c C_{p_c}} (T_m - T_c) = 0$$

where  $C_{p_c}$  is the coolant specific heat. The remaining boundary condition that was used is that the coolant temperature at  $x = 0$  equals the temperature of the coolant in the channel,  $T_c(0) = T_{ch}$ . The coolant and porous media temperatures are coupled and must be solved simultaneously. The coolant temperature can be solved explicitly, with the coolant temperature in finite difference form being

$$T_{c,i} = (d \Delta x T_{m,i} + T_{c,i-1}) / (1 + d \Delta x)$$

where

$$d = \frac{h_v}{G_c C_{p_c}}$$

and  $i$  is the nodal index that increases with  $x$ .

### Permeability

The permeability  $K$  is a function of both temperature and pressure and is defined as

$$K = \frac{B_o P_{avg}}{\mu_{avg}} + \frac{4}{3} K_o V_{avg}$$

The coefficients  $K_o$  ( $4.094 \times 10^{-8}$  in.) and  $B_o$  ( $4.96 \times 10^{-13}$  in<sup>2</sup>) are constants that are determined when obtaining the permeability as a function of temperature and pressure. The values used here were obtained from unpublished data for C/C resulting from work performed by Oak Ridge National Laboratory for NASA Langley Research Center. The average dynamic viscosity,  $\mu_{avg}$ , ( $1.039 \times 10^{-6}$  lb<sub>m</sub>/in-sec) is defined to be the viscosity at the average temperature,  $T_{avg} = 1300^\circ\text{F}$ .

The average pressure in the porous media is defined as

$$P_{avg} = \frac{P_{ch} + P_{comb}}{2}$$

The average velocity in the porous media is defined as

$$V_{avg} = \sqrt{\frac{3\kappa T_{avg}}{m}}$$

where  $\kappa$  is the Boltzmann constant ( $1.3803 \times 10^{-23}$  J/molecule K) and  $m$  is the molecular mass ( $3.34522 \times 10^{-27}$  kg/molecule).

### Mass Flux

The mass flow rate per unit area,  $G_c$ , can be calculated using an equation originally proposed by Bond<sup>4</sup>, where the temperature,  $T_{avg}$ , is the average temperature of the porous media,  $L$  is the thickness of the porous media,  $M$  is the molecular weight of the coolant, and  $R_u$  is the universal gas constant.

$$G_c = \frac{K \Delta P}{L} \frac{M}{R_u T_{avg}}$$

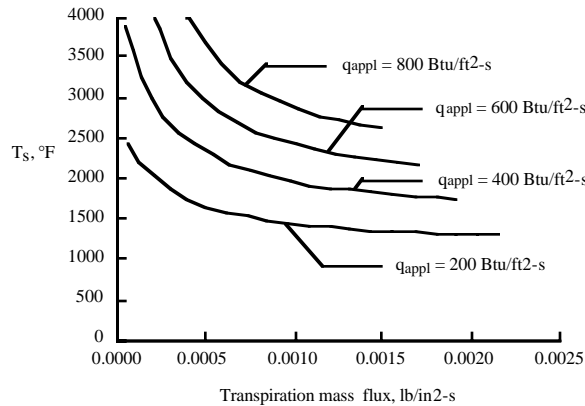
The above equation uses the average temperature of the porous media and the pressure difference between the coolant channel and the combustor. The differential equation used earlier to calculate the coolant pressure in the porous media uses the mass flux calculated here along with temperature and pressure dependent coolant properties throughout the porous media.

The porous media temperatures were solved using finite differences. They were then used to calculate the coolant temperatures and pressures. The coolant temperatures and pressures were then used to recalculate the porous media temperatures until convergence was reached.

### Results and Discussion

A parametric study was performed using the finite difference code described above to evaluate the effect of the transpiration mass flow rate (which is a function of permeability and pressure) and heat flux on the surface temperature of the heated surface. The C/C thermal conductivity and the hydrogen coolant viscosity and thermal conductivity are temperature dependent and calculated at each node. The hydrogen coolant specific heat and density are both pressure and temperature dependent. The convection coefficient in the channel was calculated numerically, and the channel was assumed to have a square cross section 0.1 in. by 0.1 in. with a mass flux of 2.55 lb/in<sup>2</sup>-s. The thickness of the porous media,  $L$ , was assumed to be 0.1 in. Since the coolant in the coolant channel was heated along the flow path, two different coolant temperatures were used, 840°F and 1560°F. The temperatures represent the inlet and outlet coolant temperatures for a typical combustor. The corresponding coolant pressures were 3648 psi and 2643

psi. These coolant temperatures and pressures are representative of pure convection cooling. The convection coefficient for the coolant in the porous media was also calculated numerically. A sketch of the geometry and nomenclature used in the thermal analysis is shown in Fig. 9.

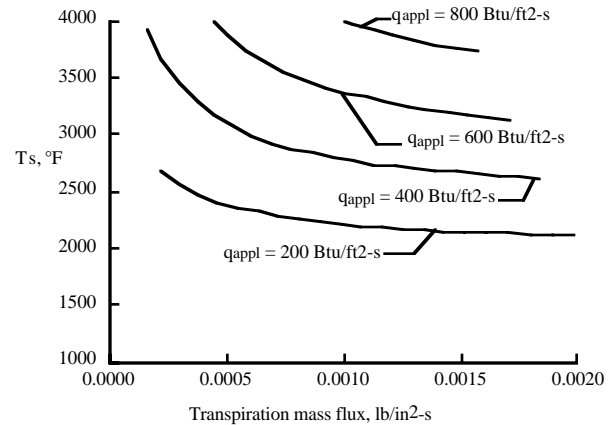


**Fig. 10. Combustion wall surface temperature as a function of transpiration mass flux and applied surface heat flux at the entrance to the coolant channel (coolant at 840°F and 3648 psi).**

In the current analysis, the surface temperature is allowed to reach a value dependent on all the conditions of the problem. The applied heat flux is given, but the surface temperature and transpiration flow rate are calculated. Iterations would then be required between the porous media solution and the boundary layer solution which computes the applied net heat flux if a coupled analysis for all three effects in Fig. 3 were desired.

In Fig. 10, the surface temperature of the C/C is plotted as a function of the transpiration mass flow rate and the applied surface heat flux for the conditions at the entrance of the coolant channel, with the channel coolant at a temperature of 840°F and a pressure of 3648 psi. The surface heat flux is assumed to include any blocking effect of the transpiring coolant. The porous media analysis does not calculate the magnitude of the blocking effect of the transpiration, but utilizes the net heat flux applied to the surface (which could be calculated by the boundary layer code). The surface temperature was calculated for four different heat fluxes, ranging from 200 Btu/ft²-s to 800 Btu/ft²-s. The mass flow rate is a function of the pressure, temperature, and permeability. Since the pressures and temperatures are determined as part of the analysis, only the permeability needs to be varied to obtain the range of mass fluxes,  $G_c$ . The permeability was varied by varying the coefficients  $B_0$  and  $K_0$ . The porosity and effective thermal conductivity of the porous media were not changed in this analysis. (Note that  $h_v$  is also a strong function of mass flux and permeability.) From the figure, it can be seen that as the

mass flow rate is increased, the surface temperature appears to become insensitive to transpiration mass flow rate, indicating that for high flow rates, the wall surface temperature essentially becomes a function of applied heat flux only.



**Fig. 11. Combustion wall surface temperature as a function of transpiration mass flux and applied surface heat flux at the coolant exit to the combustor (coolant at 1558°F and 2643 psi).**

In Fig. 11, the surface temperature of the C/C is plotted as a function of the transpiration mass flow rate and the applied surface heat flux for the conditions at the exit of the coolant channel, with the channel coolant at a temperature of 1558°F and a coolant pressure of 2643 psi. Again, the surface temperature is calculated for four different heat fluxes, ranging from 200 Btu/ft²-s to 800 Btu/ft²-s. From the figure, it can be seen that as the mass flow rate is increased, the surface temperature appears to become less sensitive to mass flow rate. Here, the wall surface temperatures are higher than for the entrance conditions, due both to the higher coolant temperature and the lower pressure, which results in a lower mass flow rate for a given permeability.

The results from Fig. 10 and Fig. 11 indicate that C/C permeabilities that result in transpiration fluxes greater than 0.002 lb/in²-s result in little further reduction of surface temperature at the given heat flux levels. The mass fluxes used for blocking in the boundary layer analysis were up to 0.0545 lb/ft²-s, or 0.00038 lb/in²-s. The mass fluxes required to reduce (block) the heat flux at the surface calculated from the boundary layer analysis are thus at levels where the wall temperature is sensitive to mass flux variations. This indicates that coupling of the two analyses would be beneficial.

## Concluding Remarks

A convection/transpiration cooling technique for cooling engine combustors was numerically studied

using both a boundary layer code and a porous media analysis. The boundary layer analysis determined that transpiration cooling can block the heat flux from the combustor. The porous media analysis indicated that values of the surface temperatures are reached where increases in the coolant flow rate (controlled by permeability) have little further cooling effect on the surface temperature. The boundary layer and porous media analysis should be coupled to further investigate the feasibility of this convection/transpiration cooling technique for hypersonic vehicle combustor walls.

## **Acknowledgment**

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## **Reference**

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